79-7013

EXPERIMENTAL EVALUATION OF DAMPING COEFFICIENT OF A SQUEEZE FILM MOUNTED BALL BEARING

V.Arun Kumar* and Pramod A. Paranjpe**

Abstract

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This paper describes a test rig built for parametric evaluation of damping coefficient as a function of mount stiffness, squeeze film thicxness, inlet oil pressure and magnitude of unbalance.

The damping coefficient is quantitatively evaluated from the direct measurement of transmitted force, damper sleeve displacement, speed and the phase angle between the transmitted force and amplitude of vibration of damper ring. Non contacting type of capacitance pick-ups were used to measure damper sleeve displacement. The transmitted force was measured by Piezo electric force transducers. Both the force and displacement signals were cross-correlated in real-time to get the phase difference between them,

The tests were carried out at different speeds to evaluate the influence of speed. The experimental results were compared with theoretical predictions to asses the reliability of theoretical formula.

Introduction

Since damping characteristics play a significant role in response and stability of rotors with large unbalances, the design of squeeze film damper requires the knowledge of the behaviour of damping coefficient with respect to various parameters.

Experimental evaluation of damping coefficient has been presented only in a few publications @ 2,3,4. Each of these has examined the effect of only a few parameters on damping coefficient. In Ref.2 the information on the behaviour of damping coefficient is not available with respect to oil pressure and at lower speeds. Though response characteristics are available in Ref.3, the damping coefficient is not explicitly evaluated under different conditions. In Ref.2,4, the variation of damping coefficient with respect to the stiffness of the rotor mounting has not been investigated.

Therefore, the objective of the present program was to carryout a systematic investigation on the behaviour of the damping coefficient with respect to mount stiffness, unbalance, oil pressures and squeeze film thickness. The damping coefficients evaluated quantitatively from the test data are presented in this paper.

Nomenclature

a	instantaneous value of damper sleeve displacement; m
A	Maximum value of damper sleeve displacement, m
B	Oil film damping coefficient, N sec/m
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ϕ	Phase angle between amplitude of vibration of damper sleeve and the transmitted force, degrees.

* Scientist, ** Head — Propulsion Division, National Aeronautical Laboratory, Bangalore-560 017, India

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Theoretical Background

The type of squeeze film damper investigated here could be thought of as a plain cylindrical journal bearing in which the journal can whirl freely but is restricted from rotation.

The load carrying capacity is given by the following equation derived by Hays[1]

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where m & n are empirical constants depending on L/D.

A purely theoretical expression has been given in reference 2 which is reproduced here

$$B = \mu L\left(\frac{R}{C}\right)^3 12\pi\left[1-\frac{\tanh(L/D)}{(L/D)}\right] \qquad (2)$$

Since the force transmitted to the bearing housing could be thought of as the summation of the forces due to the bearing spring rate and the oil film damping, it could be written as

$$f = Ka + B\frac{da}{dt}$$

Since the damper sleeve displacement is observed to be a sinusoidally varying function, the above equation may be written as[2]

$$f = KA \sin \omega t + BA\omega \cos \omega t$$

$$= F \sin(\omega t + \phi) \quad \text{Where,}$$

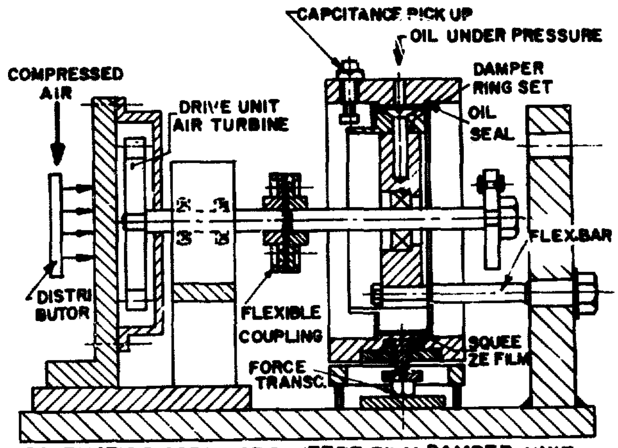
$$\sin\phi = \frac{BA\omega}{F}$$

$$\text{and hence } B = \frac{F}{\omega A}\sin\phi \qquad (3)$$

The above equation (3) wan used to evaluate the damping coefficient fron measured data. The experimental results were compared with theoretical values obtained from equation (2).

Experimental set up and instrumentation

A schematic diagram of the test rig is shown in Fig.(1). A horizontal shaft configuration was selected since it represents most of the practical applications. The experimental set up consists of a main shaft provided at one end with an air turbine which drives the rotor over a wide range of speeds. A flexible coupling was used to connect the driving and the driven units, so that any vibrations from the drive unit were isolated from the oil film damper unit.



SCHEMATIC DIAGRAM OF SQUEEZE FILM DAMPER UNIT

FIG.1

The shaft carries a damper ring assembly - consisting of an inner and outer damper ring - nounted on a deep groove ball bearing. The damper ring assembly is held by four flex bars identical to one another and cantilevered from a rigid housing. The flexible mounting of the damper ring could be varied over a stiffness value from 0.6923 x 10^7 N/m to 2,743 x 10^7 N/m in four steps by interchangeable set of flex bars. The damper ring assembly is housed in a *box* type casing, leaving a clearance all round the outer damper ring to provide an oil film.

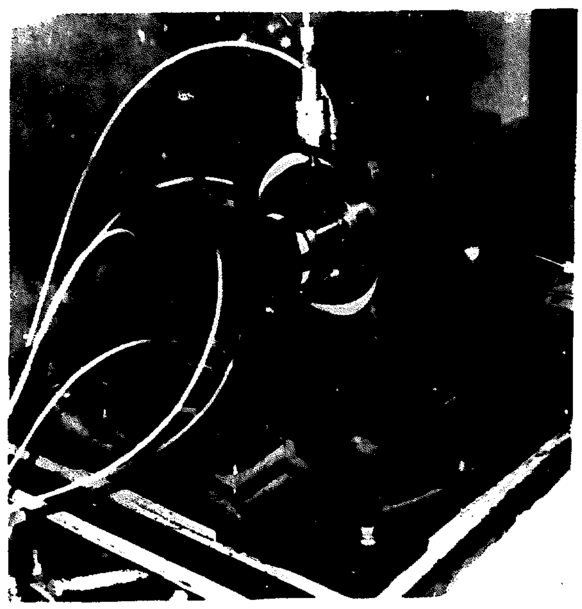


Figure:2

A circular disc mounted at the free end of the shaft carries an eccentric mass for creating the required unbalance. The test rig is designed in sucn a way that different parameters could be varied independently. The clearance space between the outer damper ring and the housing bore *is* fed with oil under pressure by using a **one** hp gaar pump. **The. whole** set up was mounted on rubber **pads** to isolate the oil film damper unit from external vibrations. A photographic view of the test rig is shown in Fig.2.

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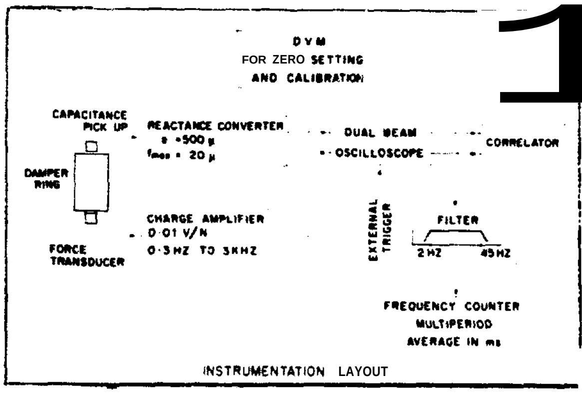
Since the force signal was little noisy **with** a signal to noise ratio of **about** 10, the force and the displacement signals were cross correlated in a real-time correlator to got the phase difference between them. This technique yielded good reproducibility in measurements. The techniqus was earlier cross-checked and calibrated ***by*** means of a variable phase generator which was wired to develop **two** signals of known phase difference adjustable from zero to 180°. **It was** found that the phase angle could be measured with an error not exceeding one degree.

The output from the displacement transducer which **is** incidentally noise free **as** compared to force signal, **was** passed through an active bandpass filter and applied to a frequency counter **for** accurate speed measurements.

Fig.3 shows the schematic of the instrumentation systen.



Fiqure: 3

Tests wsre conducted **for** four values each of mount stiffness, film thickness, oil pressures and unbalance. In each test the transmitted force, amplitude of vibration **of** the damper sleeve, rotor speed and phase angle between amplitude and the transmitted force were measured.

since signal to noise ratio was about ten in case of force signal, **it is** estimated that phase angle could be measured generally within an error of 15%. The sensitivity analysis indicates that the error in the evaluation of damping coefficient €or the above measurement error in phase angle is generally negligible. However, the error of about 10% in the measurement of transmitted force due to noise directly affects the evaluation of damping coefficient to the same extent. The other measurements were fairly noise free and accurate. In view of the **above** sensitivity analysis, it can **be** concluded that the resnlts in terms of damping coefficient would be accurate within about 15%.

Results and discussions

Although a separate ***set*** of transducers were provided for independant measurements in horizontal and vertical directions, it was unfortunately not possible to measure the forces in the vertical direction accurately due to unacceptable cross-sensitivity of the piezoelectric transducer caused by the rotor housing weight. Since this difficulty could not be overcome with minor design changes, the measurements were restricted to the horizontal direction for evaluation of the data. Therefore, the results in a sense correspond to those for a vertical rotor.

The measured data was utilised to evaluate the damping coefficient by means of equation (3). The experimental results in **the** form of damping coefficient evaluated quantitatively a5 a function of different parameters are shown in figures

4,5,6 & 7. Fig.4 shows the variation of damping coefficient with respect to pressure for three values of film thickness. It can be seen that there is a slight increasing trend of damping coefficient with respect to increase in oil pressure.

In the same diagram, theoretical values are also tabulated. It is observed that the experimental values are somewhat higher than the thsoretical predictions.

Fig. 5 indicates the variation of danping coefficient with respect to stiffness of the flex bars. It is observed that damping coefficient is not dependent on stiffness. Test values from Fig.6 do not show any definite variation with speed of the rotor, as reported by other investigators also[2].

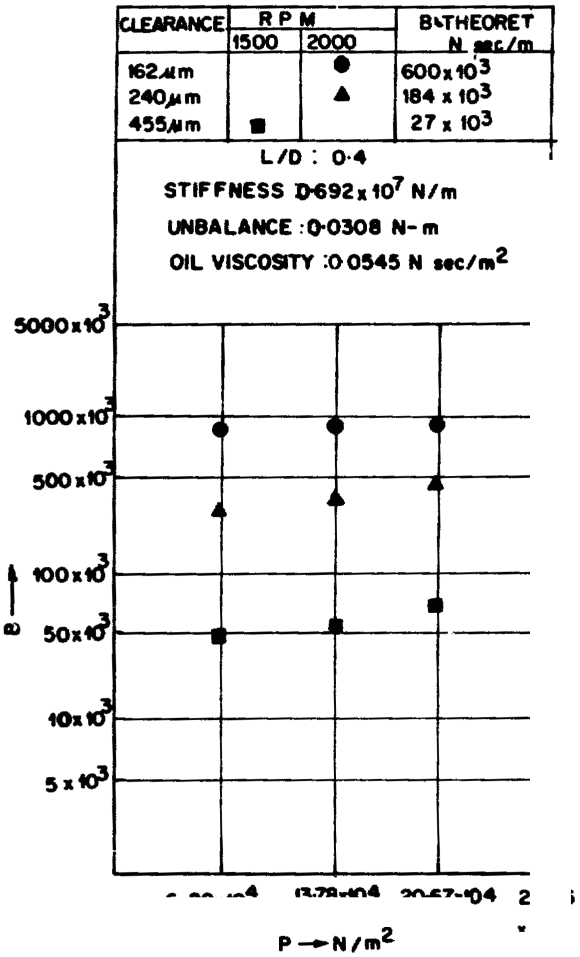
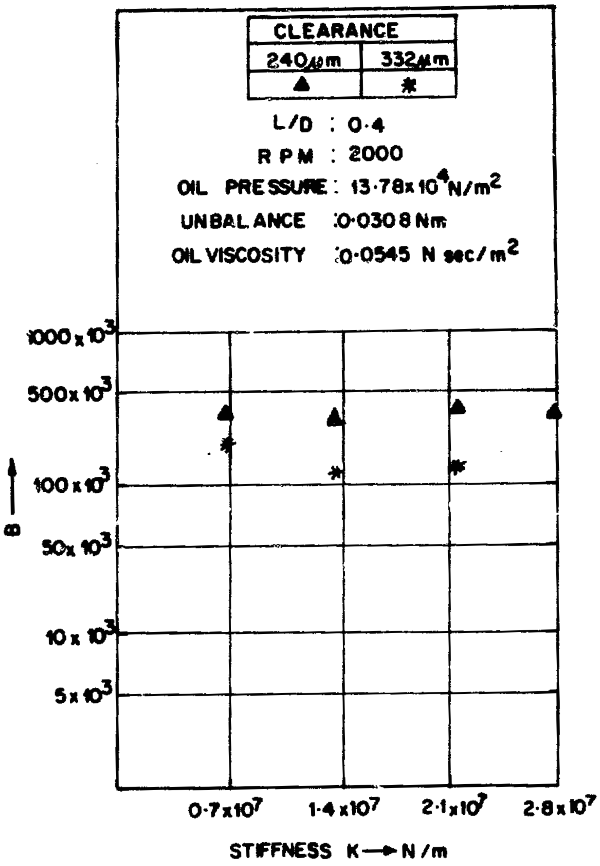


FIG. 4



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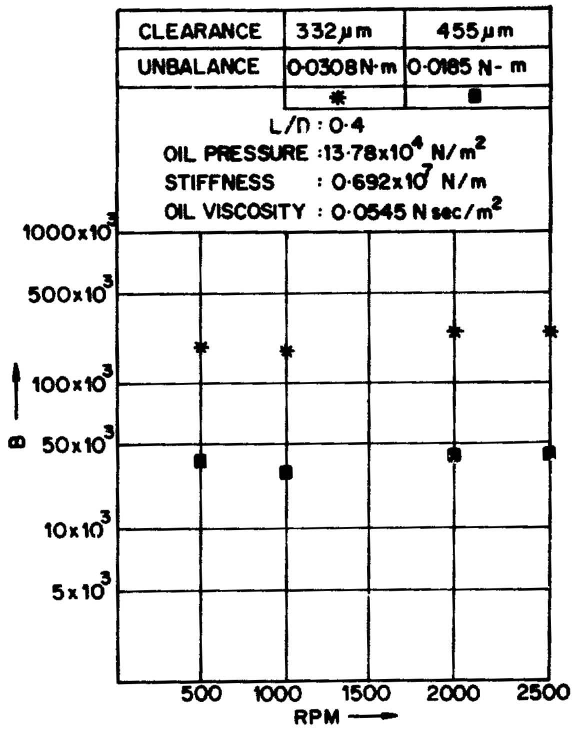
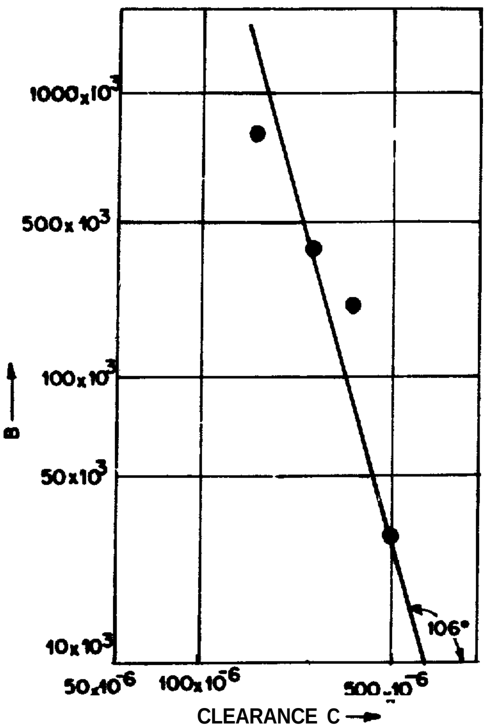


FIG 6



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The film thickness has a marked influence on the damping coefficient. Although the number of test points is small, if an attempt is made to draw a straight line through the points plotted on a logrithmic scale, the variation of damping coefficient is found inversely proportional to the clearance raised to the exponent of 3.48 as seen in Fig.7

L/D		0·4
CLEARANCE	:	$455 \mu m$
OIL PRESSURE		$13{\cdot}78 \times 10^4 N/m^2$
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OIL VISCOSITY	:	$0.0545\ N\ sec/m^2$

UNBALANCE N-m	DAMPING COEFFICIENT N sec/m
·01053	58×10^3
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Ref.3 has concluded that the damping coefficient is not amplitude dependent and hence insensitive to the amount of unbalance. The present investigations show that the damping coefficient decreases somewhat, as given in the above Table.I.

Conclusions

Experimental values of damping coefficient have been presented for four values of film thickness (162 μm, 240 μm, 332 μm, and 455 μm), four values of mount stiffness (.692 x 10^7 N/m, 1.289 x 10^7 N/m, 2.156 x 10^7 N/m and 2.743 x 10^7 N/m), 3 values of unbalance (314.64 gcm, 188.88 gcm and 107.42 gcm) and four values of oil pressures (689 x 10^2 N/m^2, 1378 x 10^2 N/m^2, 2067 x 10^2 N/m^2 an? 2756 x 10^2 N/m^2).

Experimental values are found to be somewhat higher than the theoretically predicted values of damping coefficient as also raported by other investigators[2,4]

The variation of damping coefficient with respect to speed and stiffness is found to be negligible.

Damping coefficient increases somewhat with the increase in oil pressures. The test data with respect to film thickness indicates that the damping coefficient is inverssly proportional to the clearance raised to a? exponent of about 3.48, as compared to the theoretical value of 3.

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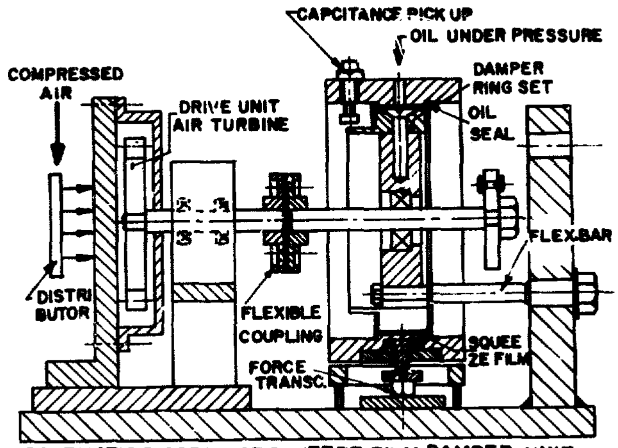

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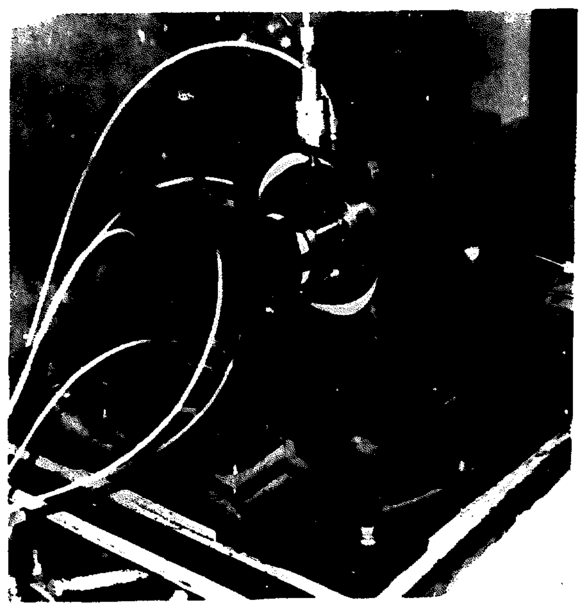

Figure:2

A circular disc mounted at the free end of the shaft carries an eccentric mass for creating the required unbalance. The test rig is designed in sucn a way that different parameters could be varied independently. The clearance space between the outer damper ring and the housing bore *is* fed with oil under pressure by using a one hp gaar pump. The whole set up was mounted on rubber pads to isolate the oil film damper unit from external vibrations. A photographic view of the test rig is shown in Fig.2.

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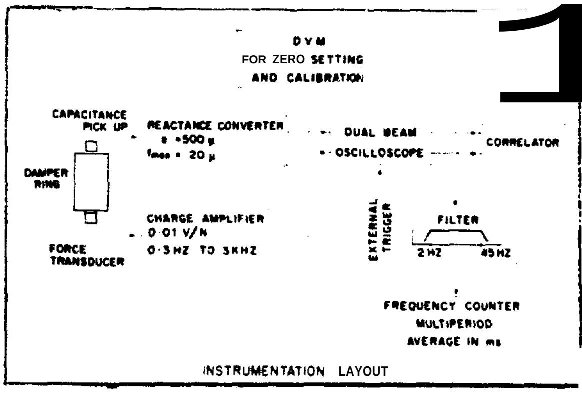

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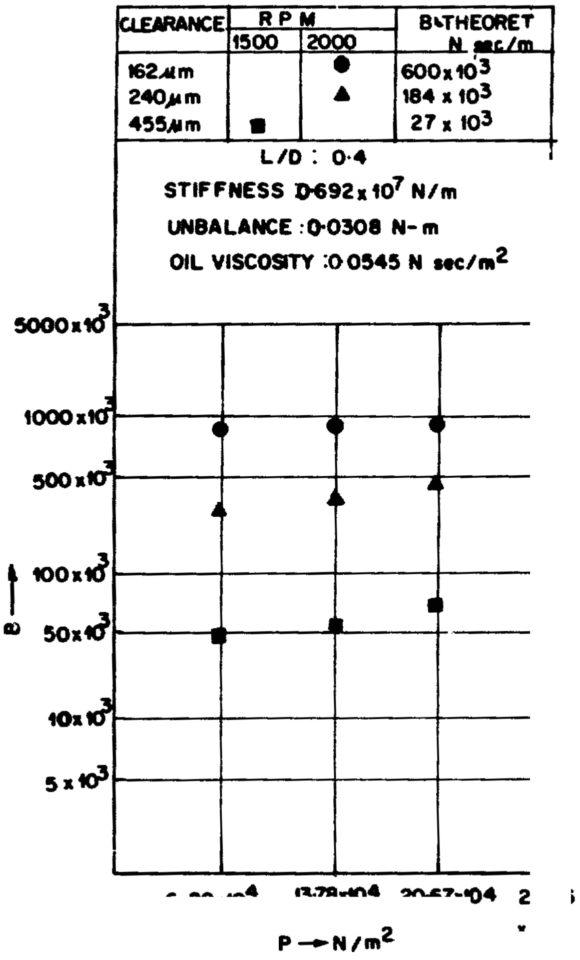

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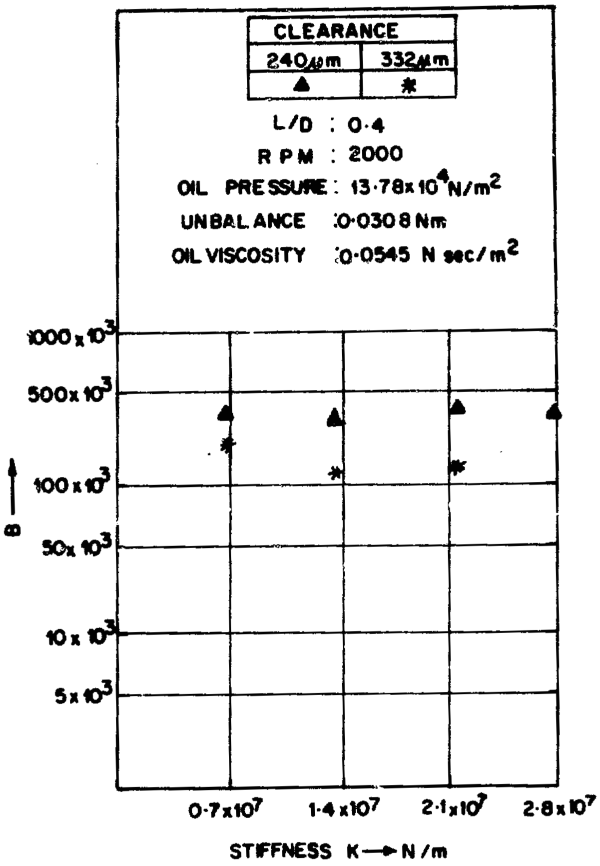


FIG. 5

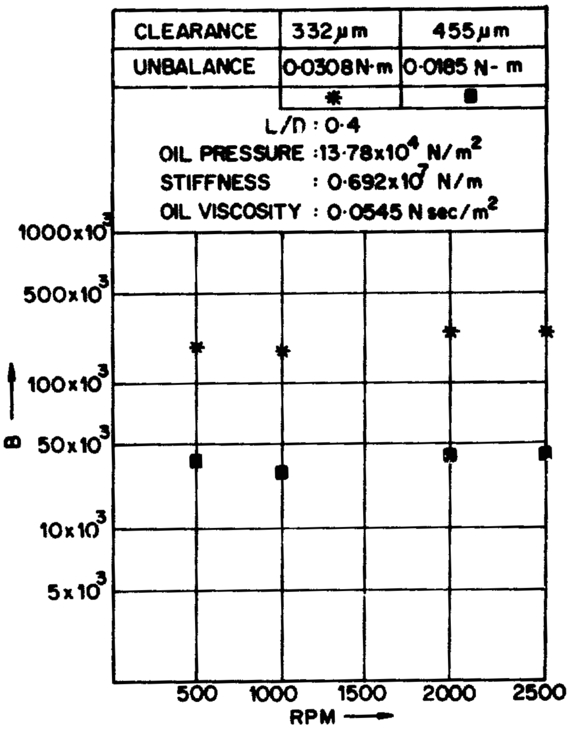
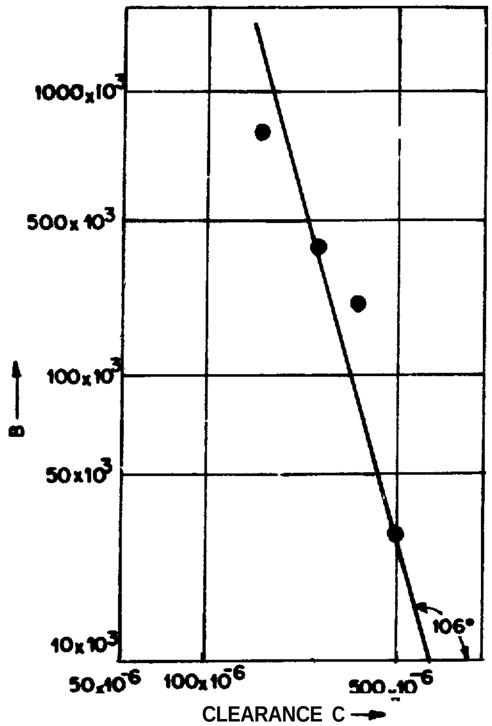


FIG 6



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EXPERIMENTAL EVALUATION OF DAMPING COEFFICIENT OF A SQUEEZE FILM MOUNTED BALL BEARING

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Abstract

Although squeeze film damper is increasingly employed as a device to control the amplitude of shaft vibrations for large unbalances or near critical speeds, quantitative information on damping coefficients over a range of geometric and operating parameters is inadequate for design purposes.

This paper describes a test rig built for parametric evaluation of damping coefficient as a function of mount stiffness, squeeze film thicxness, inlet oil pressure and magnitude of unbalance.

The damping coefficient is quantitatively evaluated from the direct measurement of transmitted force, damper sleeve displacement, speed and the phase angle between the transmitted force and amplitude of vibration of damper ring. Non contacting type of capacitance pick-ups were used to measure damper sleeve displacement. The transmitted force was measured by Piezo electric force transducers. Both the force and displacement signals were cross-correlated in real-time to get the phase difference between them,

The tests were carried out at different speeds to evaluate the influence of speed. The experimental results were compared with theoretical predictions to asses the reliability of theoretical formula.

Introduction

Since damping characteristics play a significant role in response and stability of rotors with large unbalances, the design of squeeze film damper requires the knowledge of the behaviour of damping coefficient with respect to various parameters.

Experimental evaluation of damping coefficient has been presented only in a few publications @ 2,3,4. Each of these has examined the effect of only a few parameters on damping coefficient. In Raf.2 the information on the behaviour of damping coefficient is not available with respect to oil pressure and at lower speeds. Though response characteristics are available in Ref.3, the damping coefficient is not explicitly evaluated under different conditions. In Ref.2,4, the variation of damping coefficient with respect to the stiffness of the rotor mounting has not been investigated.

Therefore, the objective of the present program was to carryout a systematic investigation on the behaviour of the damping coefficient with respect to mount stiffness, unbalance, oil pressures and squeeze film thickness. The damping coefficients evaluated quantitatively from the test data are presented in this paper.

Nomenclature

a	instantaneous value of damper sleeve displacement; m
A	Maximum value of damper sleeve displacement, m
B	Oil film damping coefficient, N sec/m
c	Radial clearance (Squeeze film thickness), m
D	Diameter of the damper ring, m
f	instantaneous value of the force transmitted, N
F	Maximum value of the force transmitted, N
K	Static stiffness of the rotor, N/m
L	Length of the damper sleeve, m
m, n	constants depending on L/D ratio
N	Rotor speed, rpm
P	Squeeze film oil pressure N/m^2
R	Radius of the damper sleeve, m
W	Load carried, N
t	time, sec
ϵ	eccentricity ratio (A/c)
μ	Oil viscosity N sec/m^2
ω	Rotor speed, rad/sec
ϕ	Phase angle between amplitude of vibration of damper sleeve and the transmitted force, degrees.

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@ the number in superscript form designates references given at the end of this paper.